Volume : 1 | Issue : 4 | April 2012 ISSN - 2250-1991

Research Paper Engineering

“Phenomenological Modeling of Four Stroke Compression Ignition Engine Processes”

* V. M. Makwana

* Assistant Professor, Department of Mechanical Engineering, Government Engineering College, Bhuj

ABSTRACT

In present work, simulation has been done for four stroke compression ignition engine for progressive combustion. Result of Progressive combustion simulation was validated with result available in literature. Graph of P-V, P-θ and T-θ were plotted for progressive combustion at stoichiometric Air-Fuel ratio. We observed that maximum temperature was 2824.49 K at crank angle in degree 381.61. Indicated power, Brake power, Indicated thermal efficiency and Brake thermal efficiency obtained from the simulation was good in agreement with the literature results. The difference between present work and literature was 2.58%, 3.11%, 5.24% and 4.73% respectively. Variations in power, mean effective pressure and efficency are presented by changing speed in graphical form. The value of Indicated power and Indicated thermal efficiency obtained from simulation of progressive combustion was 10.207 kW and 46.373%.

Keywords : Phenomenological Modeling, Progressive Combustion, Single Zone Modeling

INTRODUCTION

Ganesan, V. (2000) suggested that computer simulation can give major contribution to engine design at different levels of general studies, or detail, corresponding to a different stages of model development. In this research, the progressive combustion modeling of single cylinder for four stroke direct injection diesel engine developed considering single zone computational thermodynamic model. It is assumed that there is spatial uniformity of pressure, temperature and composition of the cylinder content at each crank angle. Combustion process considered as progressive combustion process. Air-fuel mixture is considered an ideal gas. The analysis of model covers of compression, power and expansion processes and neglecting gas exchange process during intake and exhaust. I considered the effects of heat losses, friction and temperature-dependent specific heats into performance analysis. This model can predicts in-cylinder temperatures and pressures as functions of the crank angle (θ). It can provide more realistic estimations of the main performance parameters, such as efficiency and mean effective pressure.

NOMECLATURE

C_v specefic heat at constant volume,J/kg K
N engine speed in rpm
R universal gas constant, J/mol K
t_h thickness of cylinder head, m
t_c thickness of cylinder, m
T_{ca} air temperature at cylinder head, K
T_{cw} cooling water temperature, K
T temperature, K
t time, s
U total internal energy, J
V volume, m3
W_{net} net work done during cycle, J

Greek Symbol

θ crank angle, ⁰CA
η efficency

Subscripts

1 before compression
2 after compression
3 end of combustion
4 end of expansion
5 end of exhaust

Abbreviation

bth	brake thermal efficiency
bdc	bottom dead centre
comb	combustion
disp	displacemnt
exp	expansion
ith	indicated thermal
MEP	mean effective pressure
tdc	top dead centre

THERMODYNAMIC ANALYSIS

Annand, WJD. (1963) applied more frequently phenomenological filling and emptying models for simulation of compression ignition engine. Consider the air and residual gas as control volume which is trapped between the cylinder and piston.

Mathematical Model of Compression Process

Applied first law of thermodynamics by neglecting the potential and kinetic energy (Deshmukh, N. N., & Malkhede, D. N., 2009)

$$\dot{Q} + \dot{W} = \frac{dU}{dt} + \dot{m}_d h_d - \dot{m}_s h_s \quad (1)$$

Where h_s = intake specific enthalpy, J/kg
h_d = outlet specific enthalpy, J/kg
m_s = intake mass flow, kg
m_d = outlet mass flow, kg

Assuming the cylinder walls are cooled by water and cylinder heads is cooled by air.

Heat transfer during compression process can be found by

$$\dot{Q} = A_c U_c (T - T_{cw}) + A_h U_h (T - T_{ca}) \quad (2)$$

Overall heat transfer co-efficient of walls can be determined by

$$\frac{1}{U_c} = \frac{1}{h_c} + \frac{d}{2\times K_c}\ln\left(1+\frac{2\times t_c}{d}\right) + \left(\frac{d}{d+2\times t_c}\right)\frac{1}{h_o} \quad (3)$$

K_c = thermal conductivity of cylinder material, W/m K

h_o = heat transfer coefficient at outside (water side) approximately (150 W/m² K).

h_c = inside heat transfer coefficient for turbulent flow through pipes/tubes. (Kumar, D. S., 2009)

$$h_c = \frac{k}{d} \times 0.0245 \times Re^{0.8} \times Pr^{0.4} \tag{4}$$

Where Re= Reynold number and Pr = Prandlt number

Cylinder surface area can be calculated by

$$A_c = \frac{\pi \times d \times V_{tdc}}{A_h} + \pi \times d \times s \left[\frac{1-\cos\theta}{2} + \frac{1}{s}\left(1 - \sqrt{1 - \frac{s^2}{4 \times l^2} \times \sin^2\theta}\right)\right] \tag{5}$$

Area of cylinder head can be calculated by

$$A_h = \frac{\pi \times d^2}{4} \tag{6}$$

Overall heat transfer co-efficient cylinder head can be determined by

$$\frac{1}{U_h} = \frac{1}{h_c} + \frac{t_h}{K_c} + \frac{1}{h_h} \tag{7}$$

h_h = outside (air side) heat transfer coefficient of head (20 W/ m^2 K).

Cylinder volume can be determined by

$$V_c = V_{tdc} + V_{disp}\left[\frac{1-\cos\theta}{2} + \frac{1}{s}\left(1 - \sqrt{1 - \frac{s^2}{4 \times l^2} \times \sin^2\theta}\right)\right] \tag{8}$$

Work rate can be calculated by

$$\dot{W} = -P\frac{dV_c}{dt} = -P \times w \times \frac{dV_c}{d\theta} \tag{9}$$

Where P = Pressure (N/m2) and w = angular speed, rad/s

First TdS equation for closed system (Nag, P. K. 2008),

$$\frac{dU}{dt} = mC_v.\frac{dT}{dt} \tag{10}$$

During compression process, net mass flow rate is zero hence equation (1) replaced to

$$\dot{Q} + \dot{W} = \frac{dU}{dt} \tag{11}$$

Temperature at each crank angle can be calculated by

$$\frac{dT}{d\theta} = \frac{\dot{Q} - \dot{W}}{w \times C_v \times m} = \frac{\dot{Q} - \dot{W}}{w \times C_v \times M \times n} \tag{12}$$

Hence Temperature is determined by using Euler's principle

$$T_{x+1} = T_x + \frac{dT}{d\theta} \times stepsize \tag{13}$$

Pressure can be calculated at each crank angle by

$$P_{x+1} = \frac{(N_a + N_x) \times R \times T_{x+1}}{V_c} \tag{14}$$

Where Na = no. of moles of air and Nx = No. of moles of exhaust residual gases Exhaust residual gase can be determined by

$$N_x = \frac{P_1 \times V_{tdc}}{R \times T_5} \tag{15}$$

Amount of new intake fresh air found by,

$$N_a = \frac{P_1 \times V_{bdc}}{R \times T_1} - N_x \tag{16}$$

Mathematical Model for Progressive Combustion Process

Fuel is considered as decane ($C_{10}H_{22}$) and Molecular Weight of Fuel = 142. The following are species of interest during combustion : CO_2, CO, O_2, N_2, H_2O. The fuel mass injected instaneously at tdc and enthalpy associated with pressure of injected fuel is ignored. Combustion is based on complete combustion of the fuel without delay period and dissociation. Heat released from combustion is distributed evenly throughout the cylinder. Taking stoichiometric Air-Fuel ratio, compute the adibatic flame temperature T3 at constant volume by Newton Raphson technique (Rakopoulos, C. D. , Hountalas, D. T., Rakopoulos, D. C., & Giakoumis E. G., 2005) and Np and Nr found by the applying the chemical theory. Calculate pressure that if all fuel burned instaneously at tdc by (Ganesan, V., 2000)

$$P_3' = \frac{P_2 \times T_3 \times N_p}{N_r \times T_2} \tag{17}$$

P3' =is the pressure that if all fuel burned instaneously at constant volume (tdc)

$$V_3 = V_{tdc}\left(1 + \frac{P_3' - P_2}{k \times P_2}\right) \tag{18}$$

Where k= ratio of specific heats

n= amount of fuel burned and time rate of burning is given by,

$$\frac{dn}{dt} = \frac{dn}{dV}\frac{dV}{d\theta}\frac{d\theta}{dt} = \frac{1}{V_3 - V_{tdc}} V'_{\theta} \frac{d\theta}{dt} \tag{19}$$

If the maximum burning rate is taken some hypothetical value. We can compute

$\left(\frac{dn}{dt}\right)_{\theta=\theta_3}$ where, θ_3 is the crank angle corresponding to V_3

It compared with the maximum allowable burning rate. If rate is not exceeding, then it suggest that the entire combustion process will proceed at the constant pressure. Hence

$$T_3 = \frac{P_2 \times V_3}{N_p \times R} \tag{20}$$

If the burning rate (calculated by equation), it is more than maximum allowable burning rate then whole combustion process does not happened at constant pressure and it followed in step fashion. Consider combustion happened in DN steps

$$\Delta\theta = \frac{\theta_3 - 180}{DN} \tag{21}$$

And as long as the maximum burning rate is not exceeded,

$$\Delta n = \frac{\Delta V}{V_3 - V_{tdc}} \tag{22}$$

Once the burning rate reaches maximum allowable, Take

$$\Delta n = \left(\frac{dn}{dt}\right)_{max} \Delta t \tag{23}$$

For all subsequent increment in time. The integration is completed when $\sum \Delta n = 1$ (24)

The work of expansion during combustion is calculated by

$$W_{comb} = \sum \left(P + \frac{\Delta P}{2}\right) \Delta V \tag{25}$$

Mathematical Formulation for Expansion Process

All the steps are same in expansion process as compression process but have to be started from crank angle at which combustion ends.

Exhaust temperature can found by

$$T_5 = T_4\left(\frac{P_1}{P_4}\right)^{\frac{n-1}{n}} \tag{26}$$

Where n is index of expansion

New temperature of cycle is determined by

$$T_{new} = \frac{R_c \times T_1}{R_c - 1 + \left(\frac{T_1}{T_5}\right)} \quad (27)$$

The intake valve opens at tdc and exhaust valve open at bdc and the suction and exhaust process happened at constant pressure.

Mathematical Formulation of Performance Parameter

$$W_{net} = W_{exp} + W_{comb} - W_{comp} \quad (28)$$

$$IMEP = \frac{W_{net}}{V_{disp}} \quad (29)$$

The losses of MEP due to friction in different moving parts are found by using the empirical relations. (Rakopoulos, C. D., Hountalas, D. T., Rakopoulos, D. C. , & Giakoumis E. G., 2005)

Brake MEP = IMEP – FMEP (30)

Table 1

Engine Specification	
Bore (d)	0.08 m
Stroke length (s)	0.11 m
Connecting Road Length (l)	023 m
Compression Rario (R_c)	16.5

Table 2

Expression for Simulation	
Enthalpy at temperature h(T)	$A + BT + C\ln T$
Specific heat at constant pressure Cp(T)	$B + C/T$
Specific heat at constant Volume Cv(T)	$B - 8.314 + C/T$

The coefficients for simulation are taken from given table 3 and 4.

Table 3

The coefficient A, B, and C taken for temperature range 400 to 1600 K

Gas	A	B	C	D
CO	299180.0	37.85	-4571.9	-31.10
CO_2	56835.0	66.27	-11634.0	-200.00
H_2O	88923.0	49.36	-7940.8	-117.00
N_2	31317.0	37.46	-4559.3	-34.82
O_2	43388.0	42.27	-6635.4	-55.15

Table 4

The coefficient A, B, and C taken for temperature above 1600 K

Gas	A	B	C	D
CO	309070.0	39.29	-6201.9	-42.77
CO_2	93048.0	68.58	-16979.0	-220.40
H_2O	154670.0	60.43	-19212.0	-204.60
N_2	44639.0	39.32	-6753.4	-50.24
O_2	127010.0	46.25	-18798.0	-92.15

RESULT AND DISCUSSION

Difference in power and efficiency can be arising due to difference in inlet pressure condition and in combustion efficiency. In present work, combustion efficiency was taken 100% and in literature was taken 80%.

Table 5

Code Validitation at Stoichiometric Air-Fuel Ratio, Compression Ratio=16.5 and Cut-Off Ratio =3.75

Operating Variable

	P_1 [bar]	V_1 [cm³]	T_1 [K]	P_2 [bar]	V_2 [cm³]	T_2 [K]	P_3 [bar]	V_3 [cm³]	T_3 [K]	P_4 [bar]	V_4 [cm³]	T_4 [K]	P_5 [bar]	V_5 [cm³]	T_5 [K]	IP [kW]	η_{ith}	IMEP [bar]	BMEP [bar]	BP [kW]	η_{bth}	η_m
Result in Literature	1	588.5	314.35	50.34	356.7	959.139	50.34	112.3	2572.08	6.365	588.5	1752.61	1	356.72	1216.47	10.47	43.94	14.95	12.57	8.80	36.96	84.12
Result by Present Code	1.013	589	314.59	51.127	360	962.05	51.127	111	2824.48	6.337	589	1855.79	1.013	360	1278.67	10.207	46.373	14.769	12.357	8.541	38.802	83.673
Difference in %	1.31	0.07	0.07	1.54	0.91	0.30	1.54	1.23	8.94	0.43	0.07	5.56	1.31	0.91	4.86	2.58	5.24	1.23	1.78	3.11	4.73	0.35

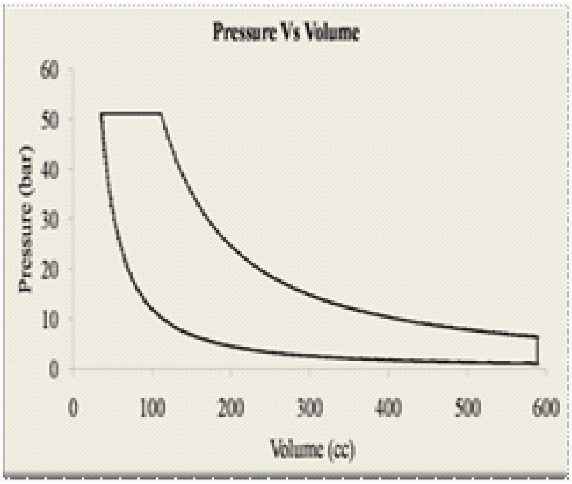

Fig. 1. P-V Diagram

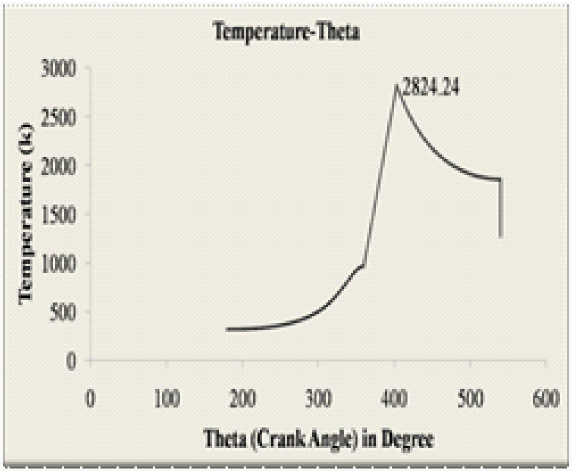

Fig. 2. T-θ Diagram

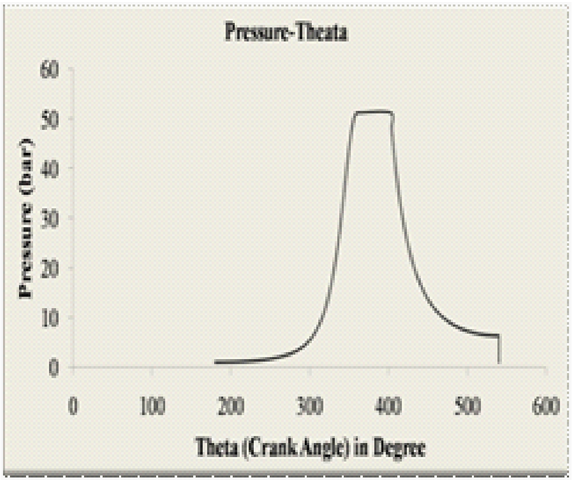

Fig. 3. P-θ Diagram

Figure 1, 2 and 3 show the variation pressure with respect to volume, temperature and pressure with respect to crank angle at stoichiometric Air-Fuel ratio, N=1500 and compression ratio=15 respectively.

Performance of 4 stroke single cylinder compression ignition engine with varying speed at stoichiometric Air-Fuel ratio

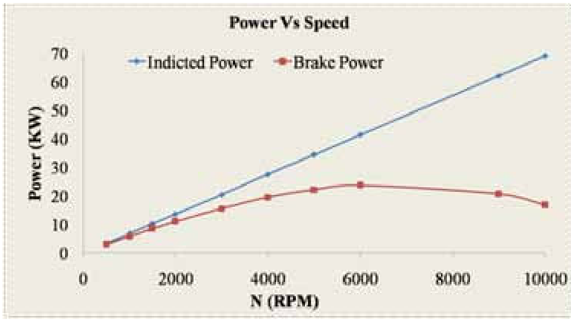

Fig. 4. Power Vs Speed

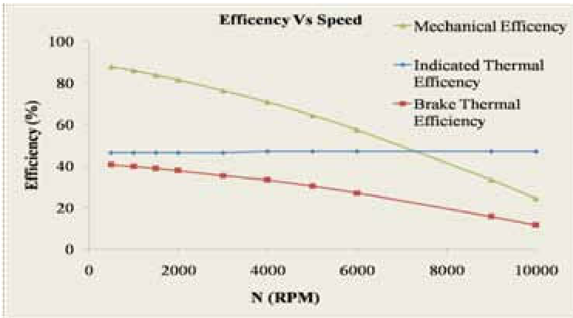

Fig. 5. Efficiency Vs Speed

According to figure no. 4, the engine speed is at 1500 RPM the Indicated Power (IP) of engine is 10.207 kW. If the engine speed is increase, IP also increases. Brake Power (BP) is increase till in 5000 rpm and then decrease and go to down because friction power increases. Maximum bp is 23.742 KW on the engine speed is 6000 RPM. .

According to figure no. 5, Indicated Thermal Efficiency (ITHEFF) is maximum equal to 47.032% and as speed increased, change in ITHEFF is negligible but brake thermal efficiency (BTHEFF) reduced. When the engine speed is increased, BTHEFF reduced from 40.593% because friction power increase.

According to figure no. 6, maximum indicated MEP is 14.977 bars at high speed. It is constant but brake MEP reduced from 12.928 bars as speed increased due to increment in the friction power and net output at engine shaft is reduced.

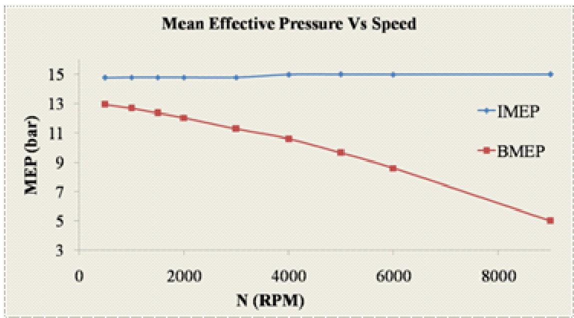

Fig. 6. Mean Effective Pressure Vs Speed

CONCLUSION AND FUTURE WORK PROPOSAL

1. The variation of specific heats reduces the temperature and pressure of the gases after the compression stroke and at the end of combustion in I.C. engine.
2. Heat transfer deteriorate the performance of the compression stroke in I.C. engine

3. Incomplete combustion of the fuel is major loss as the fuel supplied to the engine is in tiny droplets of liquid fuel. The evaporation, mixing and combustion take place inside the engine.

4. Large divergence in efficiency in present work with actual engine is partly due to valve operation, incomplete combustion and gas exchange process.

Ghojel, J., & Honnery D. (2005), Gogoi, T. K., & Baruh, D. C. (2010) suggest that progressive combustion simulation is simple, low computational cost and reasonable accurate as more complex models. This model is not however reliable, it does not cover the fluid flow analysis and do not take care of premixed and diffusive combustion

Lipkea, W. H, & De Joode, A.D. (1989) suggest that two-zone is better than single zone modeling. Quintero, H. F., Romero, C. A., & Useche, L. V. (2007) suggest four-zone or multi-zone models increased accuracy and flexibility for such complex phenomena as the formation of nitric oxide and soot in engine cylinders. Three dimensional Computational Fluid Dynamics modeling would be essential to understand the phenomenon. (Stone R., 1992)

REFERENCES

Annand, WJD. (1963). Heat transfer in the cylinders of reciprocating internal combustion engines. Proc Inst Mech Eng, 177: 983–90. | Deshmukh, N. N., & Malkhede, D. N. (2009). Experimental Performance Analysis of Single Cylinder Diesel Engine with blends of Food Grain Based and Non-Food Grain based Biodiesel, World Academy of Science, Engineering and Technology, 60. | Ganesan, V. (2000). Computer Simulation of Compression Ignition Engine Processes. New Delhi: Tata McGraw-Hill Company Ltd, University Press (India) Ltd. | Ghojel, J., & Honnery D. (2005). Heat Release Model for the combustion of diesel oil emulsion in DI diesel engines. Applied Thermal Engineering, 25, 2072-2085. | Gogoi, T. K., & Baruh, D. C. (2010). A cycle simulation model for predicting the performance of a diesel engine filled by diesel and biodiesel blends. Energy, 35 , pp. 1317-1323. | Kumar, D. S. (2009). Heat and Mass Transfer Seventh Revised Edition, S. K. Kataria & Sons. | Lipkea, W. H, & De Joode, A.D. (1989). A comparison of the performance of two direct injection diesel engines from a second law perspective. Society of Automative Engineers, Trans SAE J Engines , 98:1423–40, SAE Warrendale 890824. | Nag, P. K. (2008). Engineering Thermodynamics. 4th Edition, Tata McGraw-Hill Company Ltd | Quintero, H. F., Romero, C. A., & Vanegas Useche, L. V. (2007). Thermodynamic and dynamic analysis of an internal combustion engine with a non-circular gear base modified crank slider mechanism. 12thIFToMM World Congress, Besancon, June 18-21. | Rakopoulos, C. D., Hountalas, D. T., Rakopoulos, D. C. , & Giakoumis E. G. (2005). Experimental heat release rate analysis in both chambers of an indirect injection turbocharged diesel engine at various load and speed conditions. Society of Automotive Engineers Inc, SAE paper no: 2005-01-0926. | Stone R. (1992). Introduction to Internal Combustion Engines. London: 3rd edition, MacMillan